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TRANSPORT COEFFICIENTS OF ICE SLURRY IN PLATE HEAT EXCHANGER

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ABSTRACT

The use of ice slurry in refrigeration and air conditioning systems is relatively new. Among its advantages, it can be cited the capacity of "cold storage" in the form of latent heat and the possibility of being pumped as pure water. In order to be employed in those applications, it is necessary the knowledge of the heat transfer and fluid flow characteristics. On this project, an experimental device was developed to study the transport properties of the ice slurry in plate heat exchangers as far as the pressure drop and heat transfer characteristics. Water is used as hot fluid (thermal load). Several situations were investigated for different initial ice fractions and mass flow rates of ice slurries. The conditions of the thermal load (pure water) had been kept constant. The following variables were monitored throughout the test: the inlet and outlet temperatures (from the slurry and the water); the pressure drop, the initial ice fraction and the flow rate of the ice slurry in the heat exchanger. The results show that the cooling capacity and the overall heat transfer coefficient increase as function of the initial ice fraction and the mass flow rate of the ice slurry. Comparisons were made with water-water heat transfer cases, under the same temperature conditions. Finally, it can be showed that the pressure drop increases for higher initial ice fraction.

1. INTRODUCTION

According to the application, sensible heat thermal storage systems (cold water for example) and latent heat thermal storage systems (ice for example) are easily found on the market, each one with its advantages and disadvantages. With deserved prominence, for each system, the ease of water flow and thermal storage (latent heat) of the ice.

One of the latest technological alternatives in thermal accumulation systems consists of the production of fine ice crystals in an aqueous solution, originating the "ice slurry". It works with the good characteristics of the latent heat thermal storage systems, because the change of phase is present, associated with the capability to be pumped like a sensible heat thermal storage systems. The ice slurry systems represent an alternative with great potential to be used in systems of air conditioning and refrigeration. Ice crystals in suspension in a particular aqueous solution form the ice slurry, also known as liquid ice, "flo ice"or "ice slurry" among others. The characteristics of manipulation and refrigeration capacity can be combined for the majority of applications in refrigeration and air conditioning.

Bellas, et al. (2001) studied the thermal performance of the ice slurry in exchangers with 24 plates. The results show that the heat exchange and pressure drop of the ice slurry are functions of the viscosity of the slurry, size of crystals and ice fraction. On the other hand, Knodel, et al. (2000) observed the behavior of the ice slurry in an horizontal tube heat exchanger. They presented results for pressure drop and heat exchange coefficient for different flow conditions. The results show that, for small ice crystals, the process of heat transfer is incremented, as the friction factor. The performance of the ice slurry is evaluated for different mass flow rate by Grupta and Fraser et al. (1990), for solutions of glycol-water with 6% concentration and 0% to 20% ice fraction. The correlations for different pipe sizes were determined, showing that the heat exchange increases for small crystals of 2 to 3 mm. Sanchez et al. (2002) studied the behavior of the ice slurry, obtaining the thermo physic properties for solutions with one or several solutes. The results obtained compare solutions of ethanol- water with salt. Ticona (2003) developed an experimental device for the ice slurry generation. Heat transfer parameters in the generator were determined, as a reliable method of measurement of ice fraction when compared to other measurement methods.

2. EXPERIMENTAL APPARATUS

The experimental workbench consists of the following parts: test section (heat exchanger), thermal accumulation system (ice slurry generator, primary and secondary ice slurry reservoir, ice slurry pump), thermal load system (constant temperature bath, water reservoir, bank of electric resistance and water pump), acquisition system and the computer to store the data, observed in Figs. 1 and 2.

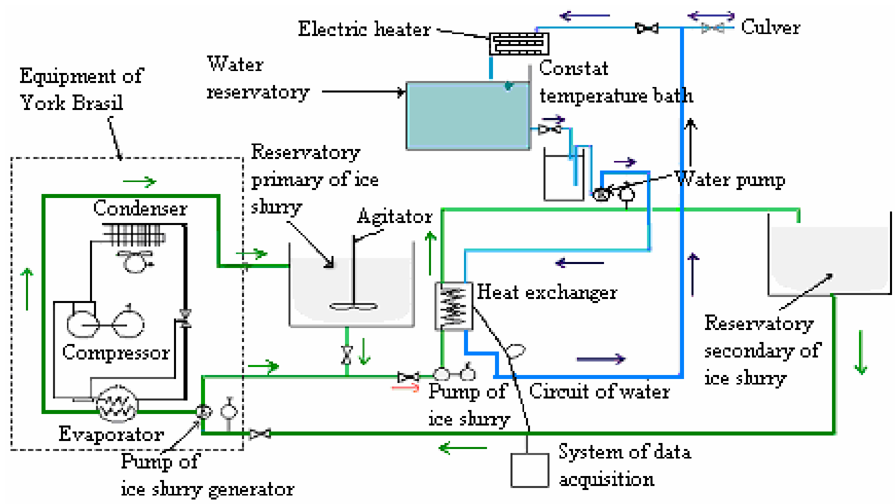

Figure 1 – Schema of the experimental workbench

2.1. Test section

The test section is the plate heat exchanger. This type of exchanger was chosen by its thermal characteristics in the application of refrigeration and air conditioning. In this project the plate heat exchanger installed was made by Alfa Laval with 16 stainless steel plates 316 and NBR as the gasket material, operating in counter flow.

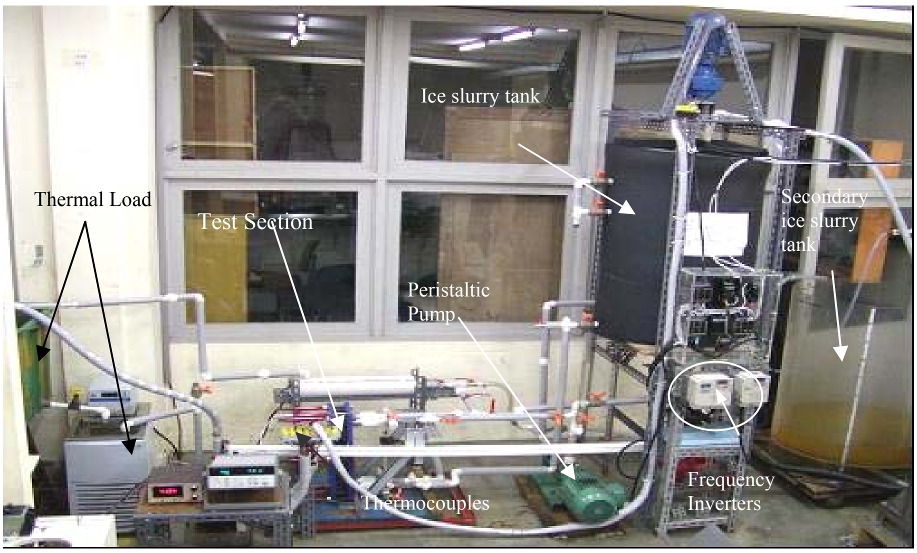

Figure 2 – Picture of the experimental apparatus

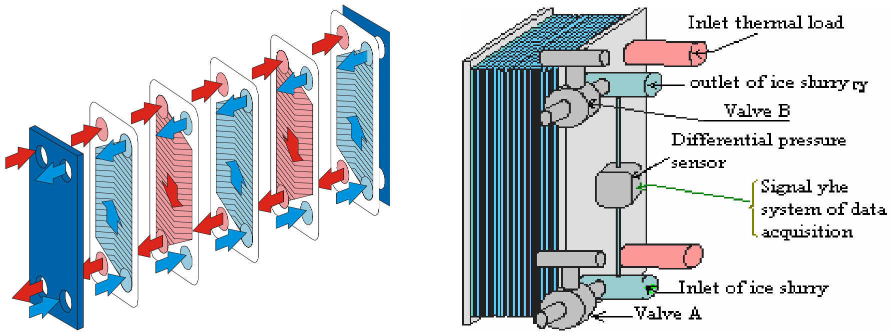

Figure 3 – Schema of the test section

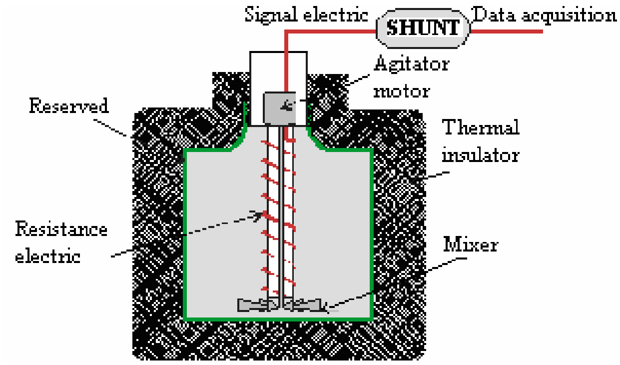

Figure 4 - Schema of the calorimeter

An exchanger schema can be observed in Fig.3. This type of exchanger allows the flow of both fluids alternatively between plates in crossed way. Each plate presents 0.032 m^2 area of heat transfer. In order to measure the ice slurry's pressure drop inside the heat exchanger, it was utilized a differential pressure transducer, in the range of 0 to 10 kPa. The voltage signal generated by the transducer is then sent to the data acquisition system.
To collect ice slurry samples, valves were installed in the inlet and outlet of the heat exchanger (A and B in Fig. 3).
For the measurement of the ice fraction on the samples, a heat meter was employed, developed by Ticona (2003), formed by an electric resistance (voltage feed of 24 V), temperature sensor and a mixer (Fig. 4). The ice slurry sample is put on the heat meter at the same time that the mixer and the electric resistance are set in motion. In the meanwhile the amperage signals, voltage and temperature are acquired for further treatment.

2.2. Thermal storage system

The thermal storage system consists of an ice slurry generator, two reservoirs, a peristaltic pump and the control panel. The secondary fluid (a mixture of ethylene glycol and water) is transported through the ice slurry generator. The refrigerant absorbs heat from the mixture generating the ice slurry that will be stored in the main reservoir where an agitator is used to keep the ice slurry in homogeneous state. A frequency inverter was used to regulate the ice slurry flow rate. The secondary reservoir is utilized to receive the ice slurry after passing by the exchanger. This way, the ice fraction in the main reservoir is not modified.

2.3. Thermal load system

This system has the function of absorbing energy from the ice slurry in the heat exchanger. It is formed by a constant temperature bath, an electrical resistance bank, a water reservoir and a water pump. The constant temperature bath (CTB) has the function of maintaining the desired temperature in the water reservoir during the tests. Three electrical resistances (2.0 kW each) heat the water, increasing its temperature close to the test temperature. Once this is done, the CTB controls the water temperature, allowing an accurate adjustment of the test temperature. The water reservoir has the function of storing sufficient liquid, in the desired temperature, to the entire test. An agitator was installed to keep the water temperature uniform. The water pump has the function of circulating the water through the heat exchanger.

3. EXPERIMENTAL PROCEDURE

The experience happens in three stages: ice slurry generation, heating of the water until reach the desired temperature and finally, interchange of heat in the heat exchanger. Ethylene glycol-water solution of 12% weight concentration is used to form the ice slurry in a continuous process. The ice slurry is then stored in the primary reservoir. The volume of ice slurry produced is of 0.5 m^3. To avoid agglomeration of crystals in the ice slurry, the mixer operates constantly. The ice slurry is obtained for different ice fractions, required in each test, from 3.0 until 25%, with temperatures between –2ºC and –3.8ºC. Parallel to the ice slurry generation, the thermal load is prepared by heating the water, previously filtered, close to 29ºC, when the resistances are turned off and the CTB starts to control the water temperature. As soon as the pre–determined conditions for the ice slurry and thermal load are reached, the data acquisition is initiated and finally the heat transfer is started. Different tests conditions, for different ice fractions and flow rates were investigated.

4. DATA REDUCTION

4.1. Ice fraction (x_s):

In order to calculate the ice fractions at the entrance and exit of the exchanger, samples of slurry are collected using the valves A and B mentioned before. Each sample of the slurry is then introduced inside a calorimeter where it will be heated, the ice melted, and the ice fraction determined. The following equations are used in this determination:

$$Q_{sen} = m_s Cp_{s'}(T_{o,s} - T_{i,s}) + m_r Cp_r (T_{o,s} - T_{i,s}) + m_{re} Cp_{re} (T_{o,s} - T_{i,s}) \tag{1}$$

$$Q_{lat} = m_s x_s L \tag{2}$$

$$x_s = \frac{Q_{el} - Q_{sen}}{L m_s} \tag{3}$$

4.2. Ice slurry's cooling capacity ($\dot{Q}_s$):

It is defined as the thermal cooling power that has the ice slurry in the heat exchanger:

$$\dot{Q}_s = \dot{m}_s [Cp_{s'}(T_{i,s} - T_{o,s}) + (x_{i,s} - x_{o,s})L] \quad [W] \tag{4}$$

4.3. Overall heat transfer coefficient (U)

The overall heat transfer coefficient is obtained as a function of the logarithm mean temperature difference (*LMTD*) and heat transfer area (*A*).

$$U = \frac{\dot{Q}_s}{A(LMTD)} \tag{5}$$

4.4. Thermal load

The thermal power is obtained in the hot side of the heat exchanger, measuring the temperatures difference and mass flow rate of the hot fluid, the water:

$$\dot{Q}_w = \dot{m}_w Cp_w \Delta T \tag{6}$$

5. RESULTS

5.1. Temperatures on the heat exchanger

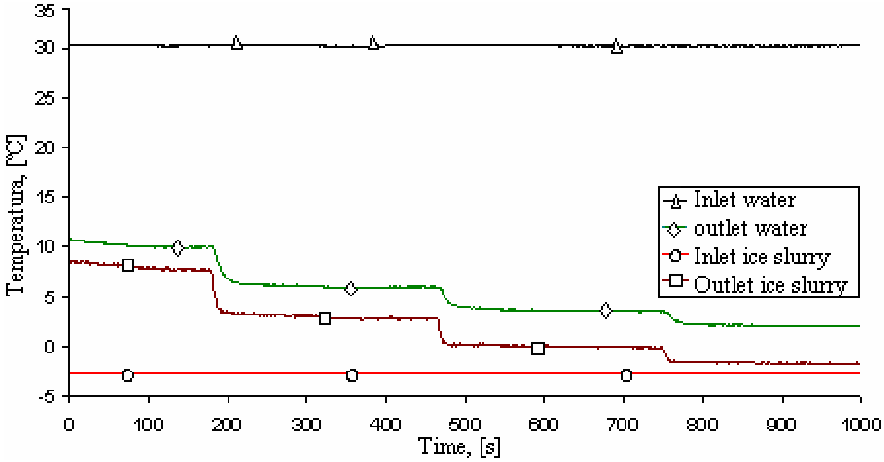

Figure 5. Typical measurement of the temperatures in the heat exchanger

Figure 5 shows the entrance and exit temperatures versus time, for both fluids, during a typical test. The inlet temperature conditions, for both fluids, are kept constant. Throughout each sequence of tests the slurry's mass flow rate is varied. It is possible to see there that, changes in the imposed slurry mass flow affect the outlet water temperature. For each test enough time is wait in order to reach the steady state, when the samples of slurry are collected for ice fraction determination.

5.2. Calorimeter

Once the sample is inside the calorimeter, the measurement of the electric and thermal parameters in the calorimeter is done. Figure 6 shows that, when the electric heater is on, the voltage drops slightly and the current goes to a constant value. With these values and the temperatures the ice fraction is determined.

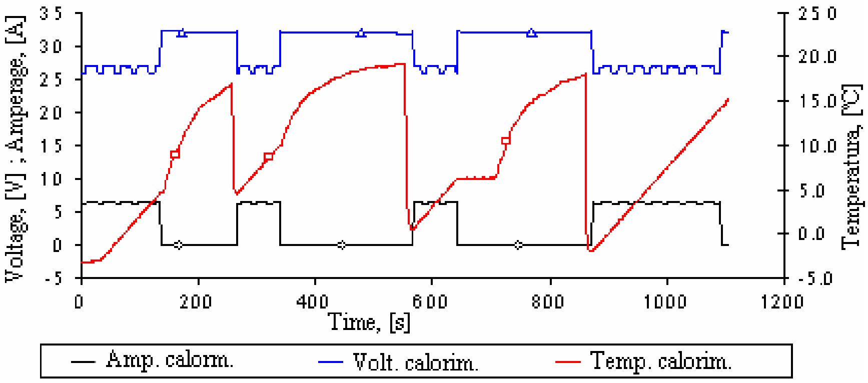

Figure 6 – Measurement of the electric and thermal parameters in the calorimeter

5.3. Pressure Drop

It is possible to see in Figure 7, an example of the pressure drop measurement. The values were acquired for different ice slurry mass flow rates. We notice that the fluctuation in the measurement increases throughout the experience, function of the flow increasing. For the analysis of the pressure drop, an average value was determined after the establishment of the steady state condition.

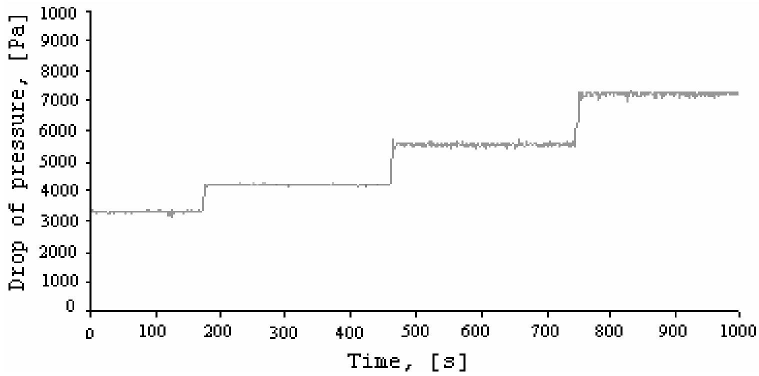

Figure 7 – Measurement of the pressure drop in the heat exchanger

5.4. Cooling Capacity

Figure 8 shows the variation of the cooling capacity of the heat exchanger in function of the slurry mass flow rate. We can observe that, for greater initial ice fractions, the cooling capacity increases considerably. When comparing with the water tests, the exchanger's cooling capacity increases almost three times . For each test the ice frction is shown in the heat exchanger's exit. In several cases the ice fraction reached "zero" in the exchanger's exit, that is, only liquid was exiting the exchanger . For greater initial ice fractions and greater flow, the fraction in the exchanger's exit is greater than zero, indicating the presence of ice crystals in the exchanger's exit.

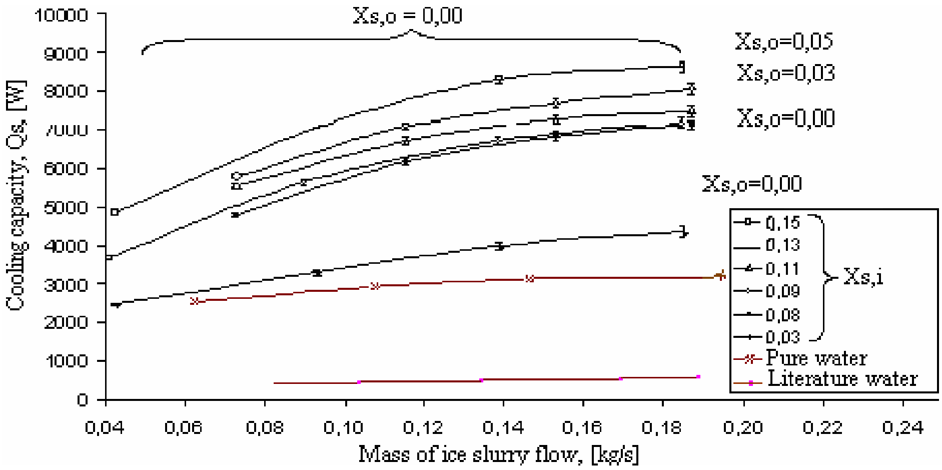

Figure 8. Cooling capacity versus slurry's mass flow rate.

5.5. Heat exchange overall coefficient

In Figure 9 s variations of the heat exchanger overall coefficient with the massive flow of the ice slurry are presented. We observe that for greater initial concentrations of ice slurry , this coefficient increases. In the same figure, we can observe the behavior of the case water-water that, as it was expected, remains very close to the case of low ice fraction. Also shown is the comparison with the Bellas'case (2001) that shows good conformity with the present work.

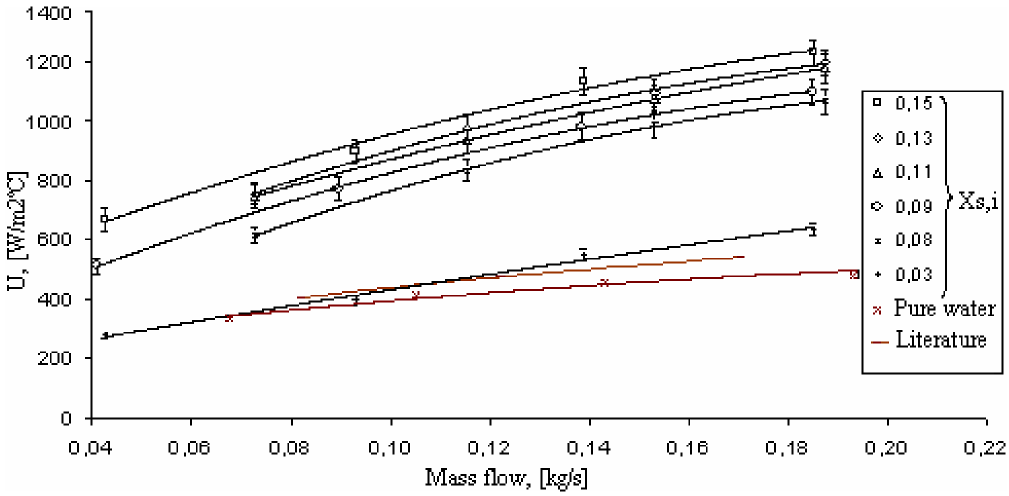

Figure 9 –Overall heat transfer coefficient versus slurry's mass flow rate

5.6. Pressure drop

We can observe in Figure 10, the variation of the pressure drop in this plate heat exchanger as function of the mass flow rate. Each case is shown for different initial ice slurry fractions. We can observe that, for greater initial ice slurry concentrations, the pressure drop increases. It is also observed, as expected, that the pressure drop increases with the flow for all different initial ice slurry concentrations. When compared to the case water-water, the pressure drop increases for the cases with ice slurry, being the case with minimum initial ice fraction similar to the pure water case.

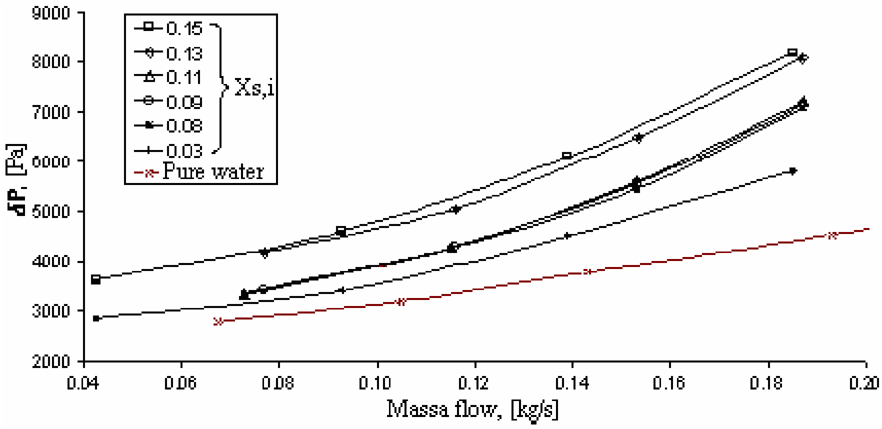

Figure 10. Pressure drop for different ice slurry flows

6. CONCLUSIONS

An experimental workbench was implemented to study the properties of ice slurry transport in plate heat exchangers. The results obtained in the present work show the technical viability of the utilization of the ice slurry through the plate heat exchanger, showing some advantages, when compared to water-water case.

The cooling capacity and the overall heat transfer coefficient increase almost three times with the utilization of the ice slurry, when compared to the case that uses only water. These advantages are accompanied by an acceptable increase in the pressure drop, which shows greater values for the friction factor.

NOMENCLATURE

Cp_{re} Specific heat of the calorimeter’s electric resistance [J/kg ºC].
Cp_{r} Specific heat of the calorimeter’s recipient [J/kg ºC].
$Cp_{s'}$ Specific heat of the carrier fluid [J/kg ºC]
Cp_{w} Specific heat of the water [J/kgºC]
L Ice’s latent heat, kJ/kg
m_{r} Apparent mass of the calorimeter’s recipient [kg].
m_{re} Apparent mass of the calorimeter’s electric resistance [kg].
m_{s} Mass of ice slurry inside the calorimeter [kg]
$\dot{m}_{w}$ Mass flow rate of water [kg/s]
$T_{o,s}$ Final temperature of the slurry inside the calorimeter [ºC]
$T_{i,s}$ Initial temperature of the slurry inside the calorimeter [ºC].
ΔT Temperature difference between the s entrance and exit the water flow [ºC].

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